GTD Magnetic Rotor Suspension Demonstrator

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***Abstract* — The use of active and passive magnetic bearings (MB) in the magnetic suspension of the rotors of the GTD is considered. Experimentally shown the occurrence of conical precession with passive MB. The directions of ensuring the required characteristics of the magnetic suspension are considered.**

***Keywords* — magnetic suspension, active magnetic bearing, passive magnetic bearing, conical precession**

I. INTRODUCTION

The use of magnetic bearings in rotor supports is one of promising trends in improvements of GTE's operational and environmental characteristics. Their use instead of roller and slider bearings gives a potential to design a "dry" GTE without an oil system, reduce friction losses, decrease vibrations caused by imbalance, etc.

Magnetic bearings (MB) are fully functional in place of conventional bearings in such sectors as high-speed machine manufacturing, power industry, medical equipment, machine-tool construction, gas transportation sector, turbomachines with power from a few kW to tens of MW, with rotor speeds from a few tens of rpm (wind generators) to tens of thousands of rpm/(high-speed spindles).

A number of demonstrators are designed abroad for aircraft engine rotor suspensions, e.g. a rotor of an auxiliary power unit (APU) with 250-kW power developed by Honeywell [1]. In the Rolls Royce's Trent 500 engine electric technology demonstrator, an active magnetic bearing is installed in the rear support of the low-pressure rotor.

Limiting factors for MB applications in GTEs are heavy weights and large sizes of the bearing and its control unit. When using in aircraft, the problem with magnetic suspension weight becomes more acute due to multiple inertial (up to 5 ... 10 g-force) and vibrational loads the engine bears in flight and landing. There are also problems with MB cooling in the hot engine section and ensuring a minimal clearance between turbocompressor rotors and MB or an engine casing.

II. ROTOR MAGNETIC SUSPENSION DESIGNING

"ERGA" R&D company in cooperation with "CIAM" developed a demonstrator of a magnetic suspension for an aircraft GTE high-pressure rotor weighing 57 kg, having 78 mm in diameter at mounting points, max. 25000-rpm rotational speed and 1.5 g permissible linear acceleration [2]. This is noticeably less than currently known designs of magnetic suspensions but exceeds by 40% the weight of suspension for the same rotor with mechanical bearings and an oil system, with an oil tank and oil pumps.

The volume of magnetic bearings is larger than the volume of mechanical ones, but they fit into the design of the engine without changing its flowpath [3]. Fig. 1 magnetic bearings 1... 10 and bearings rolling 11... 15.

Two types of magnetic bearings are known: active (AMB) and passive (PMB). The AMB requires electrical power consumption to generate and control the magnetic field - this is not necessary in the PMB owing to the use of permanent magnets.

The AMB consists of an electromechanical system or a magnetic bearing (MB) by itself, and an electronic control system. The MB contains a rotor, a stator with grooves holding windings of electromagnets, and rotor position sensors. Using information from the position sensors, this system controls the rotor position, affecting the magnetic field in the clearance by changing currents in windings of electromagnets. The control of current values makes it possible to ensure a stable centered position of the rotor, as well as to get desired values of stiffness and damping of the suspension.

The developed radial AMB is made of Grade 1311 electrical sheet steel. The AMB stator is assembled as a package having 16 poles (four poles per one electromagnet). Its outer diameter is 164 mm; inner diameter is 91 mm. Max. current density in the winding is 5 A/mm^2. The AMB sensor is an inductive transmitter transforming rotor displacements into AC voltage with specified frequency.

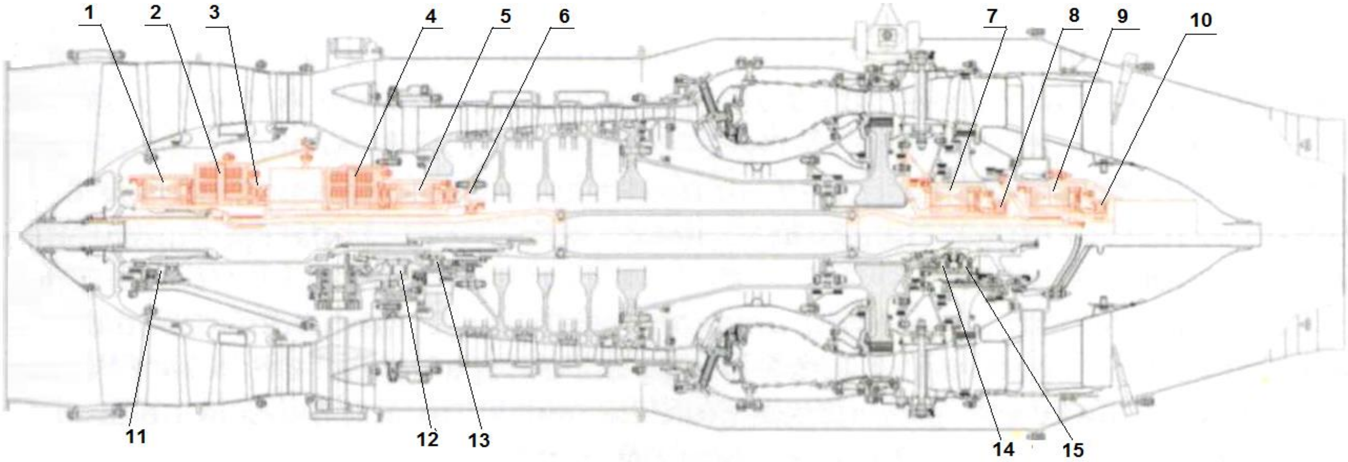


Fig. 1 Magnetic bearings and rolling bearings on the engine

The rotor suspension in the magnetic suspension with the PMB is provided due to repulsion forces of magnets on the rotor and in the casing.

A key advantage of radial PMBs is a decrease in weight as compared to AMBs with the same load-bearing capacity; their disadvantages are limited stiffness, no alignment of rotor axis in horizontal arrangements, and, as a consequence, a risk of rotor conic precession.

Individual advantages of active and passive magnetic bearings are implemented in a hybrid magnetic suspension, where radial bearings are based on permanent magnets, and a thrust bearing – on an active magnetic bearing.

The hybrid magnetic suspension should support a specified value of the clearance between the rotor and the casing at linear accelerations, vibrational loads and temperatures acting on the engine.

Basic requirements for hybrid suspension performance:

- permissible variations in the tip clearance: max. 0.5 mm
- sustained linear acceleration (g-load): ≤ 1.5 g along z-, y-, x-axis
- short-time acceleration along z-axis: max. 5.0g within 90 s.
- ambient temperature: from -60°C to +300°C

The magnetic suspension design contains radial bearings and an active thrust magnetic bearing, a rotor, safety bearings, and a control unit.

III. CHARACTERISTICS OF BEARINGS IN THE MAGNETIC SUSPENSION DEMONSTRATOR

The demonstrator of a hybrid magnetic suspension for the rotor system is developed with the following parameters: tip clearance - 1.0 mm, rotor weight - 15 kg, reference length – 450 mm, shaft diameter at bearing installation points - 78 mm, rotor speed - 0…25000 rpm; applied bearing loads: radial force - 31 *kgf* (304 N), axial force - 0…250 *kgf* (2450 N).

Fig. 2 shows a general layout of the hybrid magnetic suspension with PMB1 and PMB2 radial passive magnetic bearings, an axial AMB, an electromagnetic damper (EMD) of radial rotor vibrations and an electric drive (D). The AMB also serves as an axial damper for the PMB.

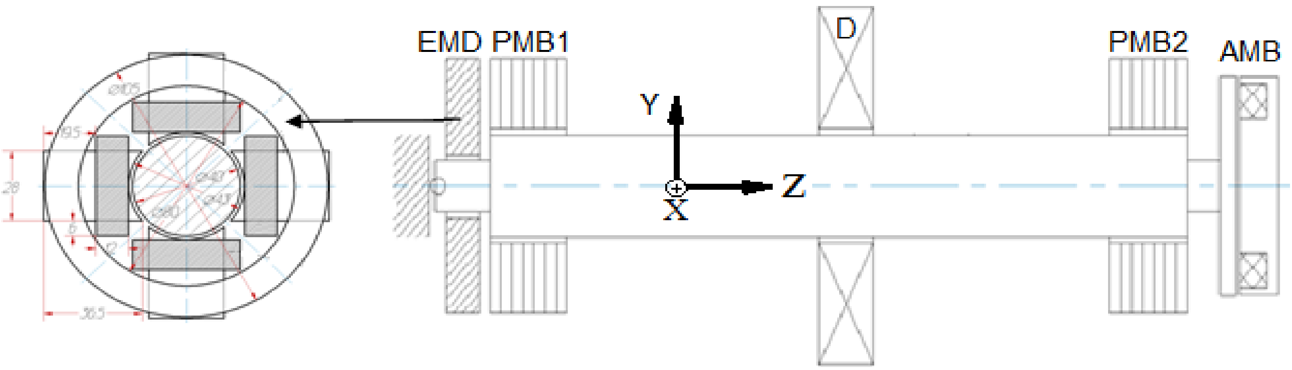


Fig. 2 General layout of the hybrid magnetic suspension for the rotor

The PMB is designed as a multilayer structure consisting of a set of rings with radially and axially oriented magnetization. The PMB geometrical dimensions are calculated by the finite element method and are shown in Fig. 3. The PMB magnetic systems are assembled “for repulsive interaction”, where an air gap between the (rotor + shroud ring) and the stator in the PMB is 1 mm.

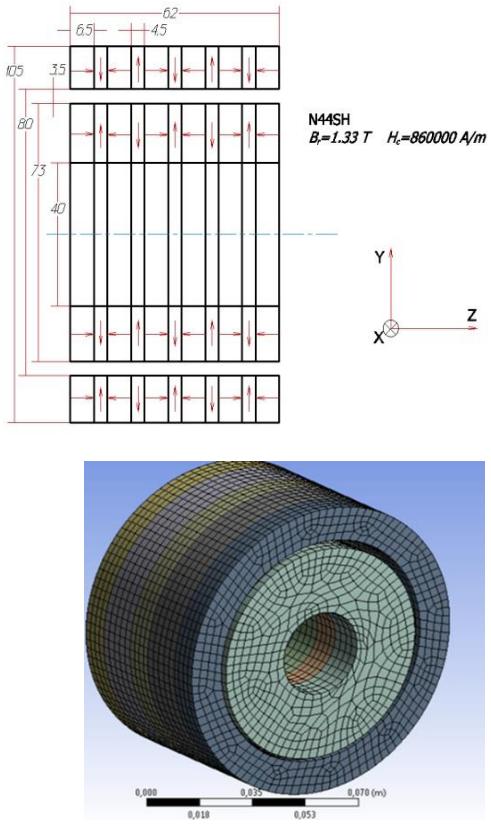
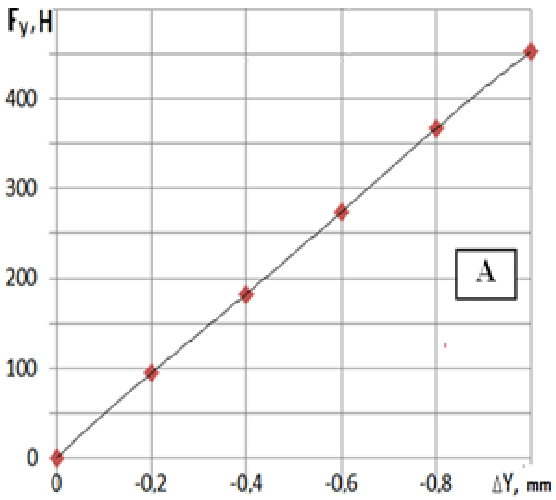


Fig. 3 Schematics and a radial PMB model used in calculations by the finite element method

The calculated dependence of the PMB radial force, Fy, on the rotor radial displacement, ΔY, without an axial displacement (ΔZ = 0) is shown in Fig. 4-A. As can be seen, the radial force increases with radial clearance variations within 1 mm, and the required value of 304 N is provided when the rotor displacement is 0.65 mm. This displacement is large enough for rotors and could be a limiting factor for the use of PMB in the magnetic suspension. The PMB radial stiffness calculated as dFy/dy is equal to 450 N/mm.



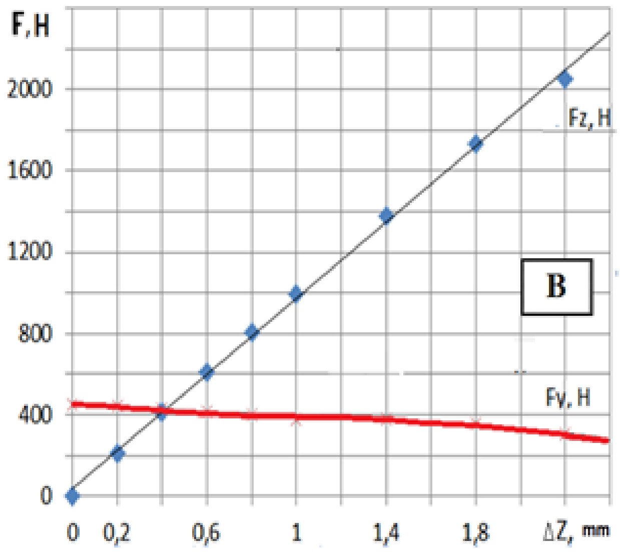


Fig. 4 Characteristics of the load-carrying capacity of magnetic bearings

Fig. 4-B shows calculated dependences of the axial force, Fz, and the radial force, Fy, acting on the rotor from the stator of one PMB on the axial displacement, ΔZ, from 0 to 2.4 mm, when the rotor radial displacement is 1.0 mm. It is apparent that the axial force, Fz, increases with an increase in the axial displacement; axial stiffness of the PMB calculated as dFz/dz has a negative value. The decrease in Fy is caused by a relative axial shift between the magnets on the PMB rotor and stator.

The partial derivative, ∂Fy/∂z, characterizes the PMB radial stiffness with axial displacements of the rotor. If its values are negative, the mode of rotor rotation could be unstable.

Fig. 5 shows calculated and experimental values of the axial force acting on the rotor in 2 radial PMBs with the rotor having axial displacements. As can be seen, when the displacement is 0.1 mm, an additional axial force equal to approx. 250 N arises. This fact should be taken into account in designing of an axial AMB. The AMB requires implementation of a control system with a high responsiveness.

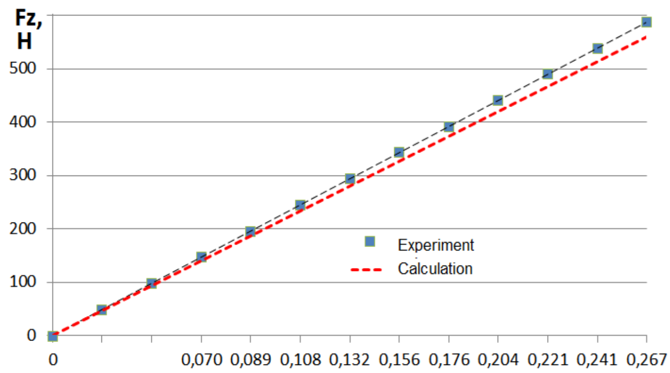


Fig. 5 Dependence of axial force on axial displacement (mm)

Fig. 6 shows photos of axial active and radial passive magnetic bearings.

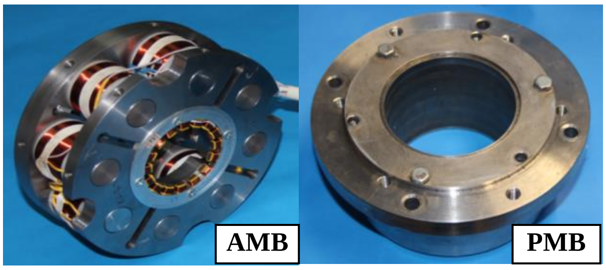


Fig. 6 Photos of active and passive magnetic bearings

Restrictions imposed on the use of passive MBs are also associated with temperature characteristics, chemical (corrosive) and mechanical properties of permanent magnets. The Nd-Fe-B magnets having the best magnetic properties are characterized by a low temperature resistance - up to 200°C, while Sm-Co magnets - up to 350°C; and the higher is heat resistance, the lower is energy characteristics. It is advisable to use Sm-Co permanent magnets at temperatures above 150°C.

IV. MAGNETIC SUSPENSION ROTOR

The magnetic suspension rotor is a multicomponent assembly and can be modified by adding or removing individual parts (Fig. 7). Its basic parameters: weight- 15kg; reference length- 450mm; diameter- 78mm.

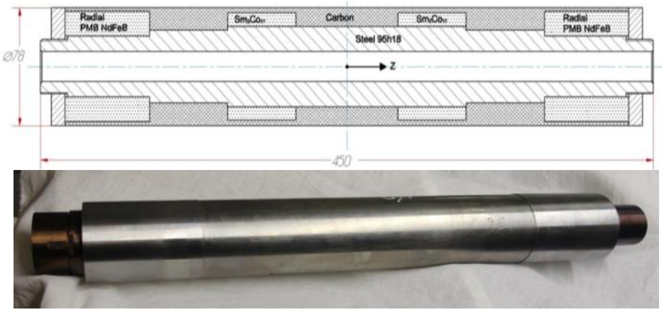


Fig.7 General view of the rotor in the magnetic suspension demonstrator

The finite element method is used in the modal analysis of the demonstrator rotor with due account of stiffness parameters of all supports - radial PMB (C, D), axial bilateral AMB (E, F) and leaf-type foil gas-dynamic bearings (A, B) used for radial damping. This rotor model is shown in Fig. 8. The mathematical model of the component is a finite element description of the structure at specified boundary conditions and loads. The following parameters are introduced: a geometry of all elements, physical and mechanical properties of materials, applied forces and attachments, contacts between parts. All parts of the rotor are taken as rigidly interconnected; contacts are specified as "welded".

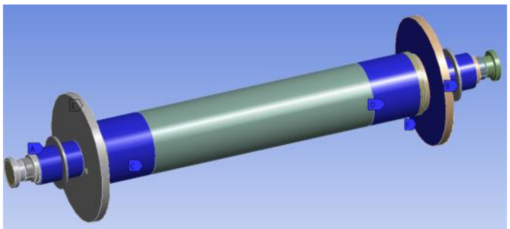


Fig. 8 Computational model of the rotor with leaf-type foil bearings
A, B – gas-dynamic leaf-type bearings (GDLB); C, D - PMB; E, F - AMB

Fig. 9 shows findings of the modal analysis - vibrational modes and their respective frequencies at 100 rpm/30000 rpm.

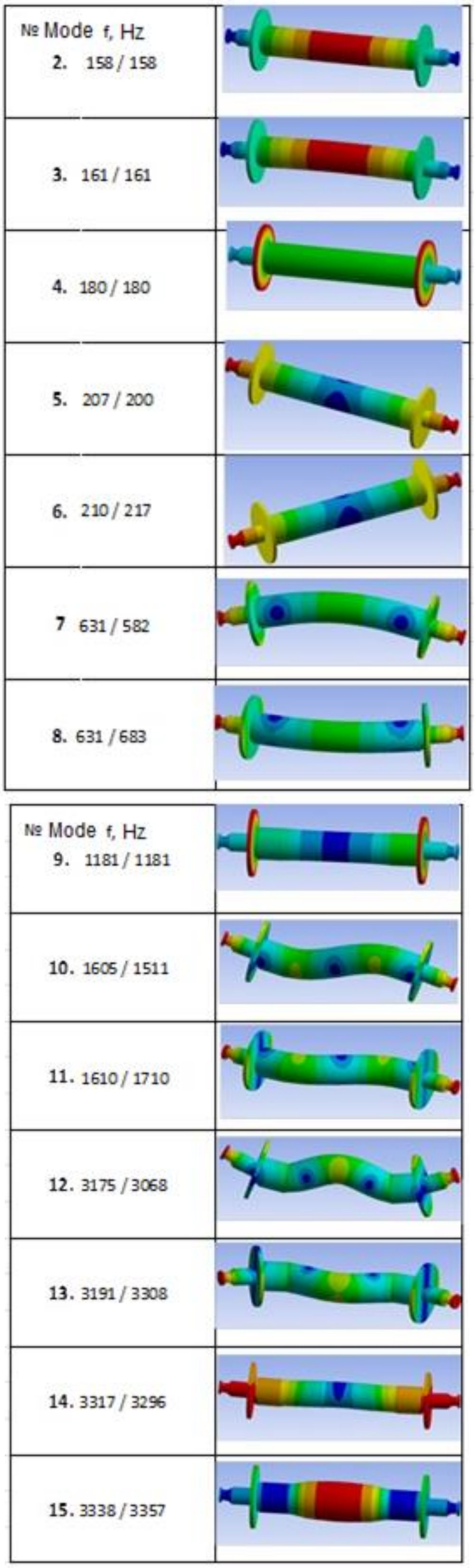


Fig. 9 Rotor vibrational modes

The rotor vibrational modes 2 and 3 correspond to the cylindrical precession of the rotor, and 5 and 6 – to the conical precession. For their passing during acceleration, additional damping of the rotor radial displacements is necessary. Other vibrations are eigenmodes, where 7-th mode (1-st flexural) has a value higher than max. operating speed and is not critical.

An electromagnetic damper (on the left in Fig. 10) or a leaf-type damper (on the right) installed in a safety bearing were studied as versions for damping radial vibrations of the rotor in the suspension structure. Calculated values of weight and dimensions of the electromagnetic damper was found comparable with the passive bearing and, consequently, it was not manufactured. Therefore, the MB demonstrator was tested with the leaf-type damper.

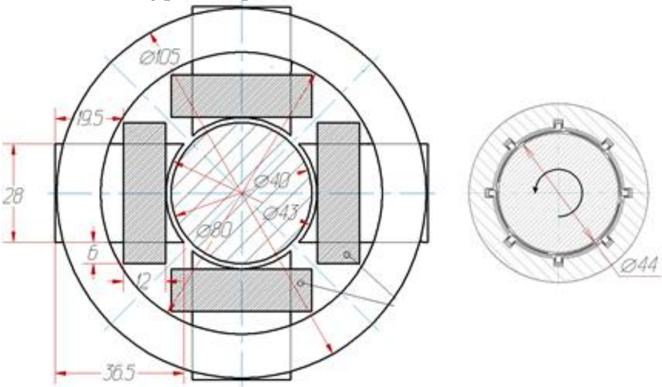


Fig. 10 Dampers for rotor radial displacements

V. TEST RESULTS FOR THE MAGNETIC SUSPENSION

Previous tests of the magnetic suspension of the rotor with the radial AMB revealed an impact on the dynamic processes in the cantilevered installation of the electric drive even with a magnetic coupling between the electric drive and the rotor [2]. The more preferable layout is the test diagram with of the magnetic suspension with installation of the rotor drive (electric drive, air turbine, etc.) between the radial bearings. Fig. 11 shows a structural layout of the magnetic suspension demonstrator driven by an electric motor with two winding zones, located between the passive radial magnetic bearings. Fig. 12 shows a photo of the control unit for the magnetic suspension.

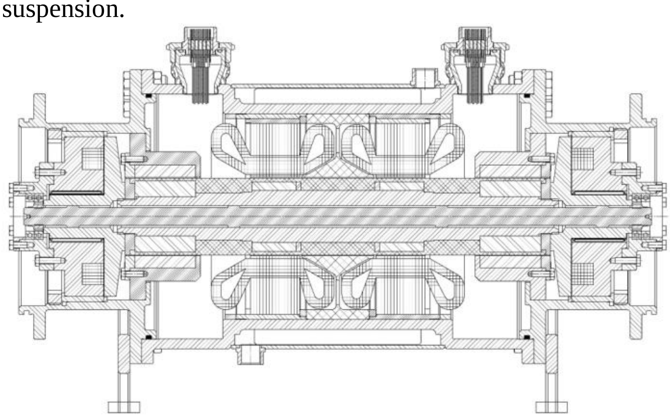


Fig. 11 Layout of the magnetic suspension with radial PMB and axial AMB

Below are the results of experimental verification of the leaf-type damper performance. Fig. 12 shows rotor precession orbits with and without the damper at 12500 rpm corresponding to the conical precession frequency. The orbits are found as the sum of signals from two mutually perpendicularly located eddy current sensors measuring the axial and radial clearances between the rotor along Y and Z axes.

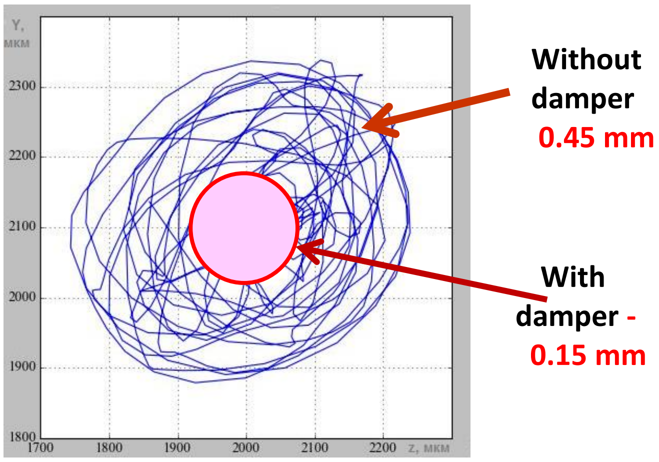


Fig. 12 Impact of the damper on the rotor precession orbit

As can be seen, installation of the leaf-type damper reduces rotor radial vibrations to one-third of its original value. At other rotational speeds of the rotor, other than frequencies of eigenmode of the rotor, its precession orbit has a shape close to a circle (see Fig. 13).

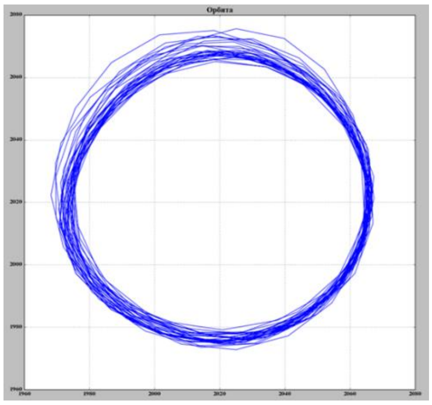


Fig. 13 Rotor precession orbit at 25,000 rpm

The leaf-type damper shows quite good performance in ground conditions, but its efficiency may decrease in high altitude conditions. It is advisable to study other designs of passive bearings with dampers, in particular, a hybrid radial magnetic bearing, combining active and passive magnets with an electromagnetic system controlling a rotor radial position.

VI. Conclusions

1. The GTE rotor's hybrid magnetic suspension demonstrator weighing 15 kg with radial magnetic bearings based on permanent magnets with 304-N permissible load and a thrust active magnetic bearing with 2450-N load was designed.

2. The use of passive radial magnetic bearings in the rotor suspension system ensures the required load-bearing capacity and supports the rotor position in the clearances relative to the bearing housing. However, without special radial dampers, radial vibrations of the rotor are possible when passing the rotational speeds corresponding to the conical precession frequency.

3. For the developed design of the GTE rotor's magnetic suspension, it is experimentally shown that radial displacements of the rotor can be reduced to one-third of its original value (from 0.45 mm to 0.15 mm) by installation of safety leaf-type bearings used also as a damper for radial displacements.

4. It is advisable to study other design versions for radial passive magnetic bearings to improve quality in controlling the clearances in bearings when passing critical rotor speeds, e.g. a hybrid radial magnetic bearing in the form of a combination of active and passive magnets.

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